**Seminar Job:** **TUTOR**
Static Modeling and Analysis Tutorial

**Topics Addressed:**

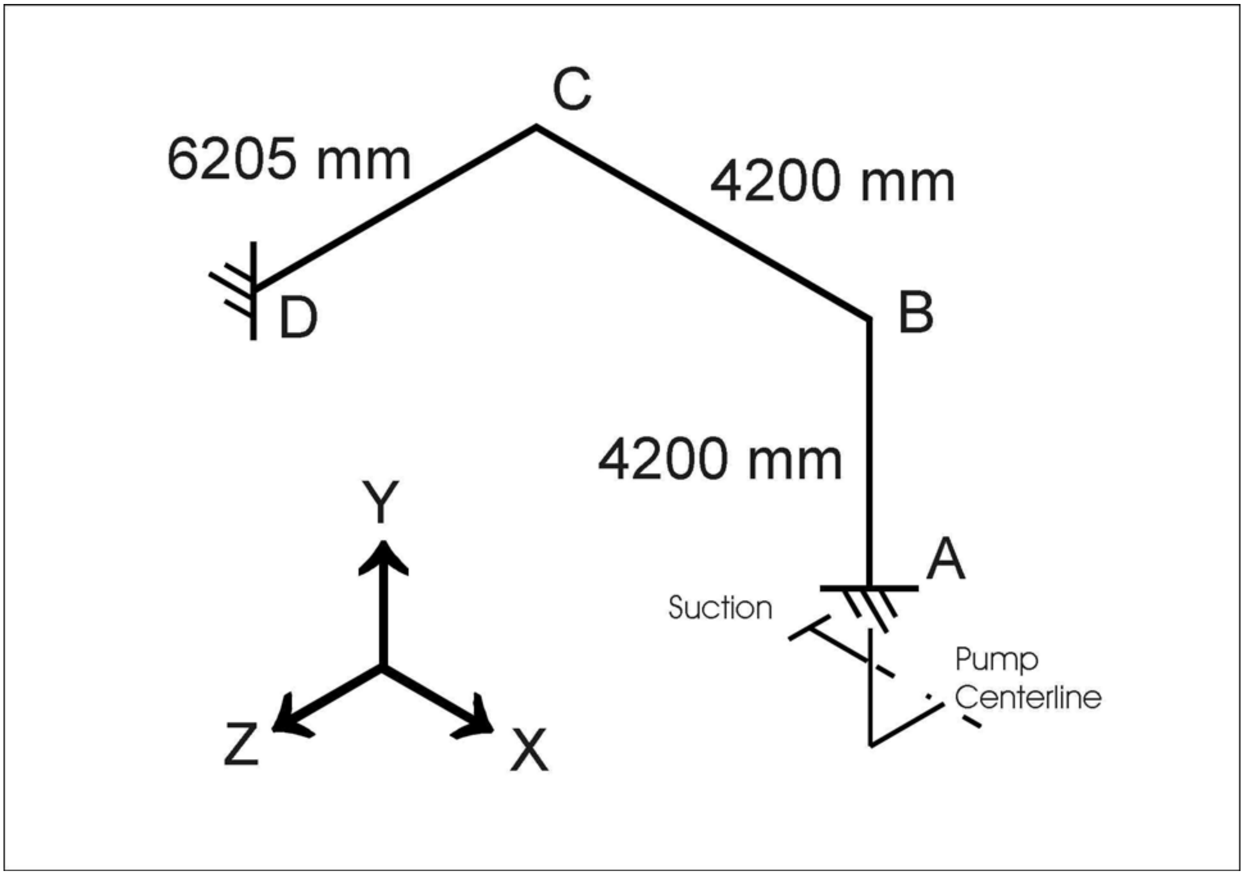
| | | |
|---|---|---|
| General Modeling | Stress Analysis | Pump Evaluation |
| Hanger Sizing | System Redesign | Structural Steel |
| Expansion Joints | Nozzle Flexibility | Local Stress Evaluation |

## Introduction:

This exercise develops the initial modeling of the system and several sequences of "run-evaluate-modify" to determine the acceptability of the system. This job reviews many of the modeling and analysis capabilities of **CAESAR II**. Starting with a quick sketch, the problem will be developed through a series of tasks, each of which will develop another aspect of the program.

## Task 1: Route pipe from pump discharge (A) to fixed nozzle (D).

With A at (0,0,0), D will be at (-4200,4200,6205)
8 inch, standard wall, ASTM A-53 Gr. B pipe
Analysis temperature = 315C
Analysis pressure = 2 bar
Corrosion allowance = 0.8 mm
75 mm C.S. insulation
Content: 0.8SG (bottoms)
Pipe specification: 150 pound class components
Use B31.3



Pump Details (A):

10 inch end suction, 8 inch top discharge
suction is -380 mm in X from pump center
discharge is 500 mm above and 300 mm in Z from pump center
Piping load on suction nozzle given as: (4450,-3550,-5340) N and
(-4070,-3390,2170) N-m

Nozzle Details (D):

Fixed end is preceded by a long weld neck flange in the -Z direction:
OD=250, wt=22, length=300 mm, weight=458 N
and a standard, 8 inch weld neck flange and gasket

Model:

Sketch & model the layout:

- After starting with a 8 inch 150 pound weld neck flange (in the Y direction) at the pump attachment (A), run up another 900 mm to the centerline of the 2 inch by-pass line (try a stub in connection—unreinforced fabricated tee). With the default node sequencing, the weld neck flange will be the element 10 – 20 and the short run of pipe will be 20 – 30.
- Place 8 inch flanged check valve 180 mm above the intersection
- Follow with another 180 mm to the second UFT for by-pass
- Continue up to nozzle elevation (B – node 70)
- Elbow to –X
- Run 4200 in –X
- Elbow to Z (C – node 80)
- Run about 5800 mm in Z
- Follow with a weld neck flange
- Finish with the long weld neck to (D – node 110)
- Add the 2 inch by-pass with its gate valve around the 8 inch check valve
  Run 450 mm in –X from the bottom intersection; elbow up, place the flanged gate valve 250 mm above the horizontal run, run pipe up to the top UFT elevation and run back to the riser

Boundary conditions:

- Set thermal growth of discharge nozzle (A – node 10)
  Two approaches: 1)Calculate thermal growth of discharge nozzle from pump base point
  Alpha =.003832, therefore displ = (0,1.92,1.15,0,0,0)
  2)Add a construction element between nozzle node and pump base
  Run a rigid element from anchored base point to discharge nozzle with appropriate material and temperature
  For this exercise specify the displacement set for node 10.
- Set thermal growth of vessel connection (D – node 110)
  Same two approaches apply – provide thermal growth of nozzle or model the vessel.
  Specify thermal growth of node 110; displ=(0,8.43,-2.87,0,0,0)
- Support riser
  Thermal growth of riser, combined with the desire to unload discharge nozzle requires that a spring support be placed near the elbow on the horizontal run (B – node 70).
- Support horizontal runs
  Suggested maximum distance between supports for 8 inch, water-filled line is 5800 mm for horizontal straight runs (see MSS SP69). Use 75% of that (4350 mm) for horizontal spans that include bends. This spacing will minimize sustained stress and line deflection thus eliminating the need for a sustained stress analysis. Since we will check these stresses anyway (and since the line weight is less than water-filled) we can exceed the suggested

spacing. Locate a restraint on each horizontal 8 inch run by using the Break command to add new node numbers – on 70-80, add node 75, placing it 1200 mm before node 80 and, on 80-90, add node 85, placing it 3000 mm after node 80. Define a (double-acting) Y restraint on these new nodes and include a coefficient of friction of 0.3. Also place a Guide at node 75 with a gap of 8 mm. Define friction for the guide.

This is job TASK1.

Results:

Code Stress checks:

- Maximum sustained stress is 11 percent of the allowable on the branch run of the tee at node 30.
- Maximum stress to allowable ratio for the expansion case is 127 percent. This value occurs on the header run of the tee at node 30.

Fix (SIFs):

One of the easiest fixes for an overstressed component is to replace it with a stronger component. Component strength is indicated by the stress intensification factor (SIF). Here, the stub-in branches are overstressed. Their in-plane SIF is 3.96 and their out-plane SIF is 4.95. Adding a pad to these tees will strengthen them. Check the effect of adding a pad by using the Tee SIF Scratchpad. Changing the UFT to an RFT with a pad equal to the pipe thickness will drop the SIF by almost 50%. With the stress here proportional to the SIF, the stress should be acceptable if the tees are changed. This modification will have no effect on the flexibility of the model. Run the analysis again with pads specified at these tee connections. (Note that a welding tee or some other self-reinforced attachment may be a better choice in light of the labor associated with attaching the pad.)

Results:

Code Stress checks:

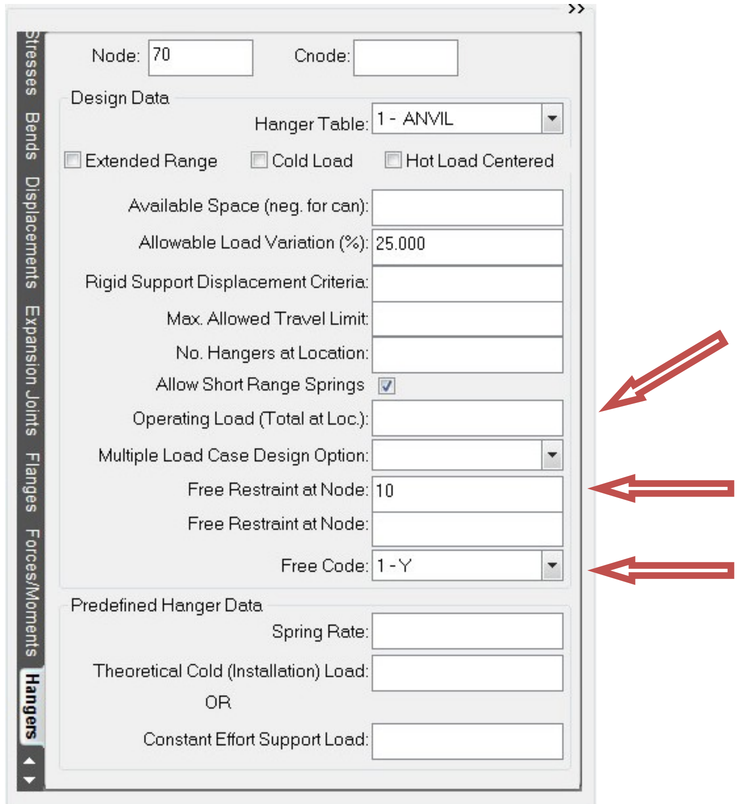
- The highest ratio of expansion stress to allowable is now at node 80 and that stress range is now 75% of the allowable expansion stress.
- The maximum sustained stress ratio is now 6% at node 60.

Hanger Sizing:

- An Anvil Figure B-268 Size 5 spring is selected at node 70
  (hot load = 1130N, deflection = 17mm, k= 11 N/mm, cold load = 1319, L.V. = 17%)
- Is this appropriate? No. Look at the operating load and installed load on the pump discharge nozzle (node 10). Typically, with a spring hanger above the pump, the pump will see a positive (up) load in the cold state and a negative (down) load in the hot state. Here, the piping pushes down on the pump in both states. This spring is undersized. Why? The calculated load carried by the spring is based on the overall distribution of weight between all vertical supports. The interaction of the pump nozzle (anchor), the spring and the other Y supports has very little load “assigned” to the hanger location. More weight is carried by the pump rather than the spring hanger.

Fix (hanger):

Deadweight that is now resting on the pump must be transferred to the hanger. The easy way to do this in CAESAR II is to remove the load-carrying capability of the pump in the initial weight analysis when the hanger load is first calculated. CAESAR II provides a convenient means to do this. On the Hanger Design data for node 70 (below) enter values for “Free Restraint at Node:” and “Free Code:” to release the “restraint” at node 10. While defined as a set of displacements, this boundary condition can also be freed for the initial weight analysis in hanger sizing. Enter “1” as the “free code”. This will release node 10 only in the Y direction. With these commands, the pump nozzle will not provide any Y support to the system when the deadweight is distributed between the proposed hanger and the other, remaining supports. Since the hanger is located very close to the riser, much of the riser load will now be carried by the hanger rather than rest on the pump discharge. Now re-analyze the system.




Results:

Code Stress checks:

- The expansion stress results are unaffected by this hanger change.
- The maximum sustained stress ratio is 9% at node 70.

Hanger Sizing:

- Now a Size 10 Anvil Figure B-268 spring is selected because the proposed “hot load” for this support has increased to 5836N (from the previous 1130N).
  (hot load = 5836N, deflection = 17mm, k= 46N/mm, cold load = 6620, L.V. = 13%)
- Review the pump load in the Y direction to determine the acceptance of this support. The resized spring now pulls up on the pump in the cold position and unloads as the system heats up. (The riser growth drops the load supplied by the spring.)
- Is this spring better? Yes. Can it be improved? Yes; even more load can be shifted from the pump up to the hanger. The hanger data spreadsheet provides for the specification of the hanger operating load (hot load). See the slanted arrow on the hanger design data window above.

Pump load:

- Review the Restraint Summary of the operating and sustained (installed) cases. This report provides no indication that these loads are excessive.
- American Petroleum Institute Standard 610 (API610) sets maximum nozzle loads for pedestal supported pumps. CAESAR II provides this calculation. Run this analysis with the pump data provided above and using the discharge loads from this analysis. Since both suction and discharge nozzles are evaluated together, the Table 4 limits in API 610 can be doubled (see API 610 Annex F).
- The discharge nozzle is overloaded. The load in the local Y direction (global Z) is excessive as are all three moment terms. The worst component is the local My (global Mz) which is over 10 times the Table 4 limit.

Fix (nozzle flexibility):

Review once again the Restraint Summary of the operating and sustained (installed) cases. These large pump loads exist in operation but not at "installation". Therefore these excessive pump loads are caused by thermal growth. These large loads can be reduced by increasing system flexibility – by either modeling existing flexibility that is not currently in the model, or, by modifying the pipe and/or support layout. Certainly the most inexpensive modification would be the provision of more modeling detail. Welding Research Council Bulletin 297 provides flexibilities for cylinder-cylinder intersections and these flexibilities might be applicable to the vessel connection at node 110.

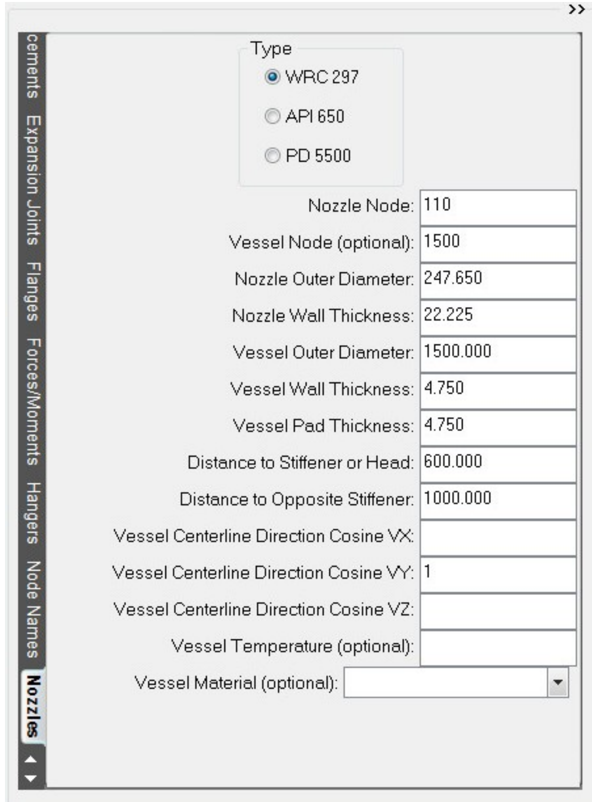
Here is the vessel/nozzle information:

- vertical vessel constructed of SA-516 Gr. 70
  - OD = 1500mm (D), wall = 4.75mm (T)
  - nozzle is 2200mm above skirt
  - skirt is 3000mm above foundation
- the long weld neck flange serves as the nozzle
  - OD=247.65mm (d), wall=22.225mm (t)
  - nozzle pad is 4.75mm. thick and 100mm. wide
  - a tray is within 600 mm of the nozzle and a stiffener ring is 1000 mm on the other side

Add Welding Research Council Bulletin 297 nozzle flexibility to the existing model.

- First, evaluate the vessel/nozzle parameters to confirm that the WRC 297 approach is valid. (d, t refer to nozzle and D,T refer to vessel shell.)
  Here, T will be the vessel thickness plus the pad thickness.
  $d/t \geq 20$: here $d/t = 11$; use this data even though it is outside the acceptable range
  $20 \leq D/T \leq 2500$; here $D/T = 158$
  $5 \leq d/T \leq 100$; here $d/T = 26$
- This flexibility will be modeled as a (zero-length) spring inserted between the long weld neck flange (100-110) and the displacement set representing the vessel growth at 110.
- Reassign the vessel growth by changing the reassigning the displacements from node 110 to node 1500. Node 1500 will not appear as a From Node or To Node on a piping spreadsheet. Instead it is referenced in the displacement set defined on the element 100-110.
- On 100 – 110 click the Nozzle checkbox to open the window to define nozzle flexibility using the data above. See below.
- Review the nozzle flexibilities listed in the error review. The nozzle provides limited axial flexibility but the longitudinal and circumferential bending flexibilities appear significant.

Reanalyze the system.




Results:

Code Stress checks:

- The highest ratio of expansion stress to allowable stress remains at node 80 but it has dropped to 65%.
- The maximum sustained stress ratio is unaffected by this change.

Hanger Sizing:

- The selected spring remains the same but the hot and cold loads increased by about 30N.

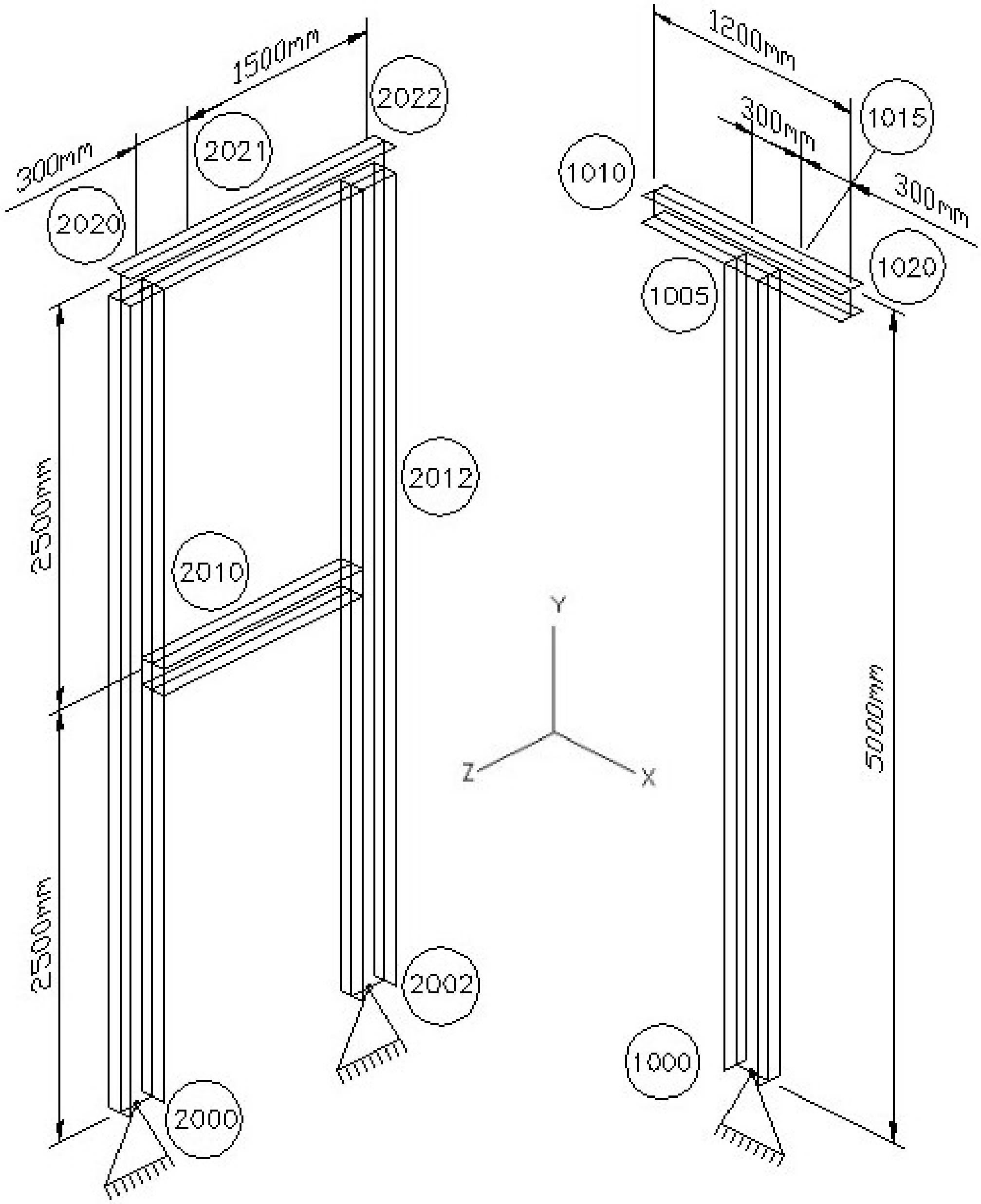
Pump load:

- A review of the Restraint Summary for the operating and sustained (installed) cases show a reduction in the operating loads on this pump nozzle.
- Running the new discharge numbers through the API610 processor shows that these loads are still excessive.

Fix (support flexibility):

There is a rather large operating load on the guide (Z restraint) at node 75. The thermal growth of the long Z run from 80-90 loads the guide and pushes the elbow at node 70 in the positive Z direction. This thermal growth increases both the pump load in Z and the bending moment about X. Is the structure guiding the pipe as rigid as the CAESAR II model says it is? If the guide has lower stiffness, the pump loads may reduce within their allowed limits. There may be reason, then, to model the structural steel that is interacting with this piping system.

There are two structures – a frame under Node 75 and a t pole under Node 85. These structures will be included in the analysis.




FRAME

UNIT EURO.FIL
MATID 1
VERTICAL=Y
SECID=1 W8X31
SECID=2 W6X20
EDIM 2000 2010 inc=10 incTo=10 last=2020 DY=2500
EDIM 2010 2012 DZ=-1800 secID=2
EDIM 2002 2012 inc=10 incTo=10 last=2022 DY=2500
DEFAULT Section ID=2
EDIM 2020 2021 DZ=-300
EDIM 2021 2022 DZ=-1500
FIX 2000 2002 by=2 all=Fixed

T POLE

UNIT EURO.FIL
MATID 1
VERTICAL=Y
SECID=1 W8X31
SECID=2 W6X20
ANGLE=90
EDIM 1000 1005 DY=5000
DEFAULT Section ID=2
ANGLE=0
EDIM 1005 1010 DX=-600
EDIM 1005 1015 DX=300
EDIM 1015 1020 DX=300
FIX 1000 all=Fixed

To build these models, start a new CAESAR II job but declare them as Structural Input rather than Piping Input. The Steel Wizard will request response through section ID before opening the general steel processor. Enter the lines of data as shown. When finished, click the Save button to process and save the data.

Now return to the piping input.

Include the steel models in the piping analysis by first clicking on the "Include structural files" button (or use the Environment menu item). Use the Browse command to select and add these two steel files to this job. Since these two steel models do not share a common node number with the piping, the program will ask where to place their local origins. Let these two models share the same origin as the piping.

Reviewing the piping and steel nodes, it is clear that pipe node 75 will be connected to steel node 2021 and pipe node 85 will be connected to node 1015. But are the centerlines in contact or does the bottom of pipe contact the top of structure? Clearly the latter is true. A more pleasing model will show the proper position of pipe and steel. And, in some cases (e.g. where friction or a guide is included on larger diameter pipe) the proper contact point will affect the results. In this system there is about 210mm between the pipe and steel centerlines. A dummy rigid element will be built at both support points to offset the pipe above the steel. First locate element 70-75. INSERT a new element AFTER this element. The new element will be from 75 to 1075 and the distance between these nodes will be -210 mm in Y. Make this a weightless, rigid element. Now return to the element 70-75 and redefine the restraint at node 75 by changing the Node to 1075 and define a CNode (connecting node) of 2021. Be sure to do this for all restraints at this point. The plot will now reposition the Frame under node 75. Follow this procedure for the restraint at 85, creating a new element 85-1085 and connecting 1085 to 1015.

Reanalyze the system.

Results:

Code Stress checks:

- The highest ratio of expansion stress to allowable stress remains at node 80 but it has dropped to from 65% to only 38%. Why? Flexibility has been added to the system. Specifically, the stiffness of the rigid guide at 75 is now in series with the frame stiffness. The 8 mm gap is closed but then the guide itself shifts another 8+mm in –Z. Compare the Z displacement of nodes 1075 and 2021.
- The maximum sustained stress ratio has a minor increase.

Hanger Sizing:

- The selected spring remains the same with a minor reduction in hot and cold loads.

Pump load:

- A review of the Restraint Summary for the operating and sustained (installed) cases show a significant reduction in the operating loads on this pump nozzle.
- Running the new discharge numbers through the API610 processor shows that only the moment about the global Z axis exceeds the limit.

Adding the steel effectively increased the guide's gap. This greatly reduced the pivot action and the resulting pump load.

Fix (modify layout):

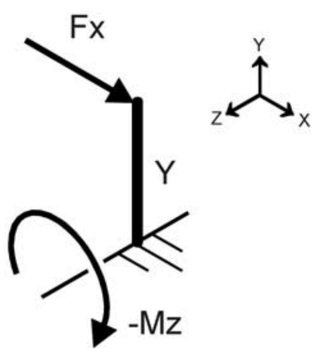
Without changing the position of the pump or vessel nozzle, or changing the thermal strain; the only way to reduce these loads is to add flexibility to the layout. There is no inherent flexibility that was (conveniently) excluded from the model so an expansion loop will be introduced.[1]

How big a loop is required and where should it be placed?

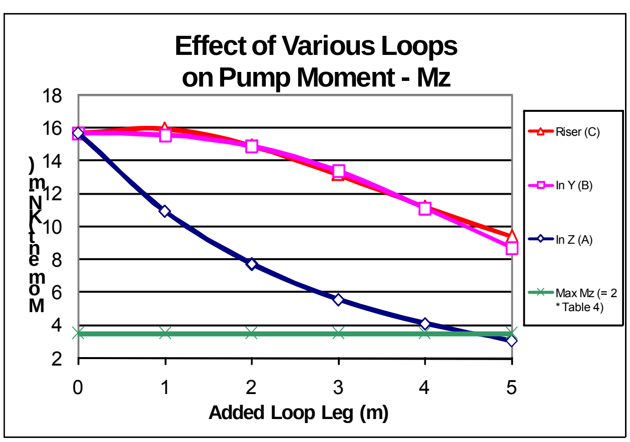
[1] One other item in this model that has not been evaluated is the sensitivity to friction effects at these supports.

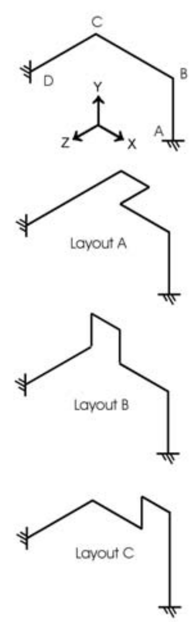
## Task 2: Design an expansion loop.

- Added legs of loop should be laid perpendicular to the thermal growth causing the load (See page 2-28 of the course notes.)
- Attention will be focused on the most excessive pump load component. The API610 report shows that this is the moment about the local Y axis. This is the bending moment about the global Z axis.
- Determine which element's strain causes this high moment about the Z axis
    - o What force acting on the Y cantilever leg causes a negative Z moment in the nozzle?
    - o +FX cross Y gives a negative Z moment at 10
    - o The thermal growth of the X run causes the negative Z moment




- The added loop legs can go in the Y or Z direction (perpendicular to the strain in X).
- What is the most effective orientation and location?
    - o A Z loop on the "C end" of the X run (Layout A)
    - o A Y loop on the "C end" of the X run (Layout B)
    - o A Y loop on the "B end " of the X run (Layout C)
    - o A Z loop on the "B end" of the X run and a Y loop on either end of the Z run is the same as Layout B
- Run through bending moment equation (LOOP LEG.XLS)
    - o Page 2-28 of course notes shows the bending stress at the nozzle is estimated as $SE=6ER\Delta L_j/\Sigma(L_i^3)$
    - o Use this to evaluate the change in moment by changing the flexible legs
    - o SE=M/Z=MR/I
    - o $M=6EI\Delta L_j/\ \Sigma(L_i^3)$; $6EI\Delta$ is constant; let $6EI\Delta=K$
    - o Solve for K using current M and Current L's
    - o Now calculate M as L changes for each design



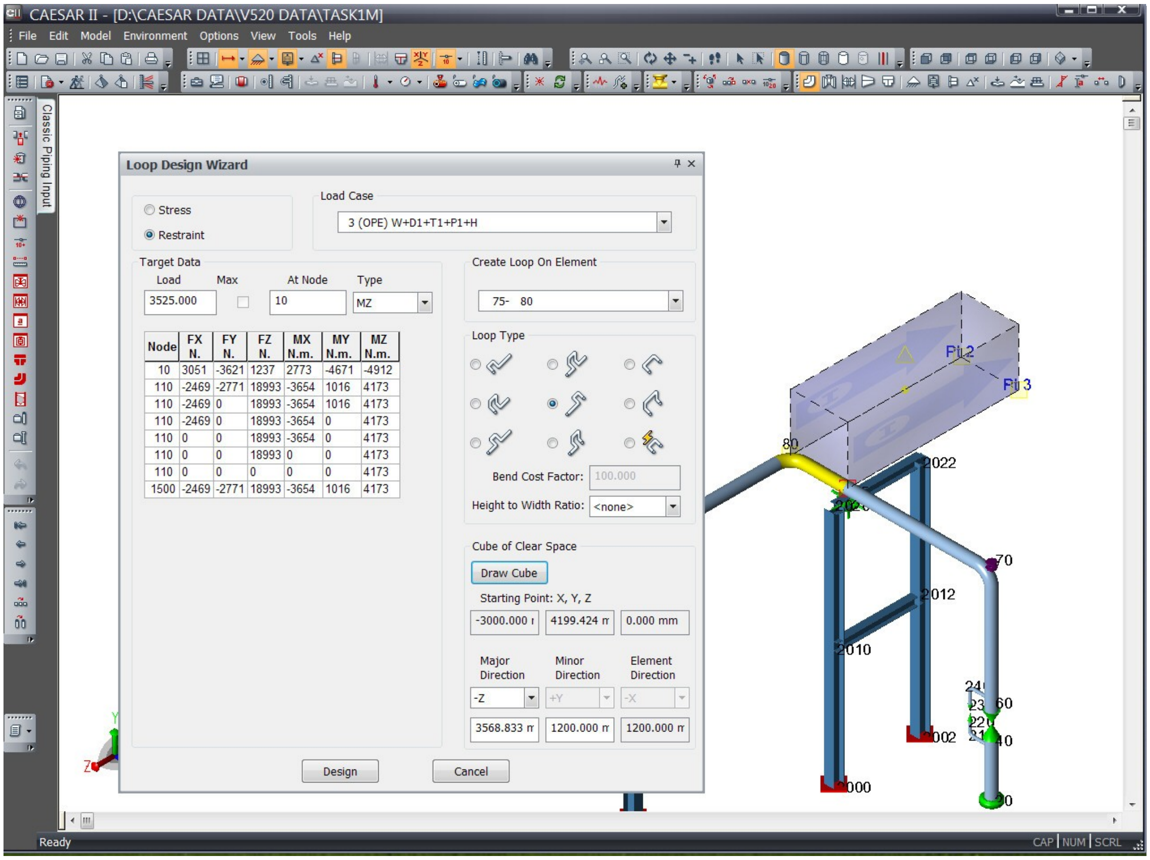



- Sketch A is most effective indicating about a 4500 mm run in Z
  Why? It makes the long leg longer and this value is cubed in the denominator.
- While the advantage of the loop in Sketch A is clear, the estimated effect is quite conservative as the simple equation does not consider the rigid elements or elbows.
- Also, the equation used above ($6ER\Delta L_j/\Sigma(L_i^3)$) does not consider any intermediate supports such as the guide at node 75.

Model:

Add loop using the Optimization Wizard:

- Return to the input for TASK1 and select (or highlight) element 75-80 as this is where the loop will be placed.
- Click the Optimization Wizard button or select this from the menu item – Model.
- Reposition the piping system and the wizard window so that both are visible.
- (See illustration below)
- The wizard will use existing results in modifying the layout. So select the operating load case from the drop list under "Load Case".
- Click the Restraints radio button to display the restraint loads for this operating case.
- In the restraint load report double-click on MZ for node 10 to load the Node and Type into Target Data.
- Enter 3525 (N-m) as the Target Data Load. This is twice the API 619 Table 4 local Y moment for an 8 inch nozzle.
- The loop will start immediately after the support at 75, offset in –Z and go out in –X to the long Z run and then continue in Z to the current node 80. This configuration is similar to the Loop Type in the center of the available Loop Type selections. This loop starts with a straight run and adds a flat loop a the end. Click this radio button.
- Select <none> for the Height to Width Ratio so that the loop will start immediately after the elbow is completed after node 75.
- Use the Draw Cube button to indicate the space available for the loop. Click and drag the corner labeled "Pt 3" in the –Z direction until the Major Direction cell lists –Z and the length in the cell below registers over 3000mm.
- Click Design to start the search for a loop in -Z that will meet the limit of 3525Nm for an operating load about Z at node 10.




Clean up and reanalyze the model:

- The Wizard sizes a loop that extends less than 1000mm in –Z. (Recall that the simple hand calculation projected an added length of over 4000mm.) It may be sensible to round this length up to an even meter and drop the pump load a little further. Do this by changing the length of 76-77 and 78-80 to 1000mm in the appropriate directions.
- The support configuration should also be reviewed. Is there sufficient deadweight support to carry the added pipe? Will there be excessive sag in the loop? Does the loop interfere with other components?
- Rather than adding another support, the sustained stress and the pipe sag will be monitored in the results. The loop's interference with the frame's space will be ignored here.
- The node number sequence between 75 and 80 did not provide enough "room" to assign consecutive node numbers around the two new bends. To include the near- and mid-points on the new elbows just de-select and re-select a bend at each of these locations (76 & 77).
- Once complete, it will be necessary to review all results for this model.
- Rename this new model by selecting the menu item File, then select Save As and save this new model as TASK2.
- Run the analysis.

Results:

Code Stress checks pass:

- Maximum sustained stress is 11 percent of the allowable at the hanger node
- Maximum stress to allowable ratio for the expansion case has dropped to 35 percent and is now at the end of the elbow starting the long Z run.

Hanger Sizing:

- A Size 10 Anvil Figure B-268 spring is still the selected support with a minor reduction in hot load.
  (hot load = 5701N, deflection = 17mm, k= 46N/mm, cold load = 6472, L.V. = 14%)

Pump load looks much better

- Rerun API610 check with new data to confirm
- (Global) Mz is 1.98 times the allowable.
- Other Appendix F checks (validating the 2 timed Table 4 approach) also pass.

Conclusion:

- The Z moment is 99 percent of its allowable limit (1.98/2.0). How much confidence do we have in this calculation?
- What is an easy way to reduce the Y load on the discharge nozzle? Will doing so improve the overall loads? How does this affect the confidence in the pump loads?
- Should the loop be extended?

## Task 3: No room for a loop; install an expansion joint instead.

- Select an 3.5 kg/cm$^2$ class, 8 inch expansion joint out of the Senior Flexonics / Pathway catalog.
- What sort of joint is needed here? Review the types of joint assemblies.
- A tied expansion joint on the riser and below the valve will be best suited to absorb the horizontal pipe growth over the pump. The vertical loads associated with thermal expansion can be adjusted by the spring at Node 70.
- Have CAESAR II calculate the free horizontal growth of the joint by breaking the system above the pump. Use this value to select the number of convolutions. Then install the expansion joint and analyze its suitability.

### TASK 3a – Determine the service requirements (travel) for the expansion joint assembly

Model:

Open TASK1 and immediately Save As TASK3

Break the run above the pump:

- Change the From Node of the element 20 – 30 to 21. The relative horizontal displacements of nodes 20 & 21 (between elements 10 – 20 and 21 – 30) will be close to the required deflection of the expansion joint.
- (The plot will have the two sub-systems sharing the same origin.)

Reconnect 20 and 21 in the appropriate directions:

- Add a restraint in the Y direction at 20 with 21 as its connecting node (CNode)
- Add the three rotational restraints between 20 and 21
- (Leave the transverse directions , X & Z, free)
- (The plot once again is fine.)

Results:

Using the expansion case displacements, calculate the change in position between Node 20 and Node 21.

- Nodes 20 & 21 move together in Y, RX, RY, & RZ because of the NODE/CNODE restraint definitions
- Delta X is 34 mm and delta Z is 11 mm resulting in a relative horizontal displacement of 36 mm

Check the pump operating loads.

- The Z moment on the pump is 4800 N-m
- This is a very high load for a “zero stiffness” expansion joint. Why is it so high?

The catalog shows a 20 convolution joint provides 38.8 mm of lateral deflection

- But it also adds a lateral stiffness of 6 kg/mm or 58.7 N/mm.
- This stiffness, if modeled, would reduce the deflection.
- Reduced deflection drops the required number of convolutions and, in turn, increases the stiffness between 20 & 21
- Model the stiffness, check the deflection, select the joint, and model the stiffness until the deflection test fails or the pump load components become too high.

Iterate to find the minimum required number of convolutions

- Add those final two restraints, in X and Z, between 20 & 21 with a stiffness set to 58.8 N/mm

- Relative lateral displacement = 22 mm. The global moment Mz on the pump is only 877 N-m but the moment about Y is excessive at over 6000 N-m. This moment will also place torsion on the expansion joint and this torsion may be excessive as well.
- If this observation did not stop the iteration, how would this process proceed?
  - Test 16 convolutions - 16 convolutions allow 24.8 mm lateral deflection and has K = 118 N/mm
  - Reset X and Z restraint stiffness to 118 and reanalyze
  - Check travel limits for the proposed joint and the load limits for the pump.
  - If 16 convolutions are OK and overall joint lateral translation drops, test a shorter (i.e., fewer convolutions) joint

Conclusion:

- Because of the large global My on the pump and the torque on the expansion joint, the proposed length and location of this joint should be reconsidered.
- For this exercise, though, analysis of a 20 convolution, tied expansion joint will be evaluated (for this length a tied universal joint would probably be preferred; consult manufacturer for other options)

## TASK 3b – Model and evaluate the 20 convolution expansion joint assembly

Model:

- The flanged expansion joint would be located between the discharge nozzle and the existing weld neck flange. To save time in this examination, the expansion joint will be placed between the flange and pipe rather than between the nozzle and flange. The error introduced will be small.
- Once again open TASK1 and immediately 'Save As' TASK3
- Move to the pipe element 20 – 30
- Enter Expansion Joint Modeler
- Select a 50 pound class, tied, 20 convolution tied expansion joint with slip on ends
- Place the joint at the From end (20)
- Adjust stiffness to pipe temperature
- Note details:

| Non-concurrent Movement | | | | Spring Rate | | | | Length | | |
|---|---|---|---|---|---|---|---|---|---|---|
| axial | lateral | angular | torsion | axial | lateral | angular | torsion | bellows | overall | weight |
| 87.9 | 38.9 | 9 | 0.212 | 69 | 59 | 9 | 7102 | 317.5 | 406.4 | 412.8 |
| | | | | 73 | 62 | 9 | 7504 | :adjusted to 315 C | | |

Results:

Using the expansion case displacements, calculate the lateral deflection between Nodes 21 and 22, which bound the expansion joint.

- Delta X is 21 mm and delta Z is 12 mm resulting in a lateral displacement of over 24 mm. A quick check of the catalog shows that this works for a 20 convolution joint
- There is minimal axial deflection and angular rotation. Torsion in the joint is 0.33 degrees. This is excessive.

Check the pump loads

- Operating loads do not look unusual. Sustained (installed) loads are small indicating that these operating loads are due to thermal growth.
- Running the API 610 report shows that this layout is acceptable with the largest load component being the global My at 0.98 times Table 4.

Evaluate the expansion joint:

- Estimate the number of expansion cycles for the life of this joint at 2000.
- Run through the linear interaction formula for a quick check

| | Actual | Allowed | Ratio |
|---|---|---|---|
| Axial | 0.34 | 87.9 | 0.004 |
| Lateral | 23.8 | 38.9 | 0.612 |
| Bending | 0.0005 | 9 | 0.000 |
| | | Sum of ratios: | 0.616 |

  - o Confirm the twist is within its limit: actual = 0.3368 deg. & allowed = 0.212 deg.
  - o JOINT FAILS
- Run through the EJMA check

| | Actual |
|---|---|
| Axial | 0.34 = X |
| Lateral | 23.8 = Y |
| Bending | 0.0002 = Theta |
| Eff. Dia. | 239.6 = D |
| Flex. Length | 317.5 = L |

X+0.00872665*D*Theta+3*D*Y/L = 54.22

- These checks are also available in CAESAR II

Conclusion:

There is too much rotation (torsion) placed on this expansion joint.

Otherwise, a 20 convolution expansion joint will safely provide the added flexibility required for proper pump operation.

Complete the Expansion Joint Specification Sheet (Appx. A of the Standards of EJMA)

Other convolution counts are available; watch out for fatigue (rating of 2000 cycles); consult the manufacturer.

## Task 4:  Evaluate the nozzle loads on the attached vessel

Model:

Use the Welding Research Council Bulletin 107 processor available from the CAESAR II Main Menu/Analysis

- Evaluate the vessel/nozzle parameters to confirm that the WRC 107 approach is valid
  $d/D \leq 0.3$; here $d/D = 0.165$
  $Dm/T \geq 50$; here $Dm/T = 157$
- Vessel and nozzle data is listed in Task 1 above.
- The nozzle vector must point to the center of the vessel for proper load conversion
- Collect the sustained and expansion loads from TASK2

Calculate local stresses using WRC 107 but use the stress summations from ASME BPVC Section VIII Div. 2 Appendix 4 – Design Based on Stress Analysis.  See also WRC Bulletin 429 – 3D Stress Criteria Guidelines for Application.

- Push the button
- Pm (general, primary membrane) stress has yield strength as its limit thus ensuring no failure by gross distortion.  This is away from the junction discontinuity and is simply calculated using pressure stress equations. Pm < Smh
- Pm+PL (primary membrane) stress is an indicator of excessive plastic deformation.  This stress combines the local membrane stress (stress that is constant across the cross section) due to sustained loads (from WRC 107)  with the pressure term in Pm. Pm+PL<1.5Smh
- Pm+PL+Q  (primary plus secondary) stress monitors shakedown.  These primary plus secondary stresses are used to monitor fatigue. Q is calculated using the WRC 107 results—bending stress from sustained loads and membrane and bending stress from expansion loads.   Pm+PL+Q<1.5(Smc+Smh)
- There is no Pb stress in this evaluation.  Pb is bending due to pressure.  This is monitored just as Pm but the limit is 1.5Smh to account for the shape factor.
- Additional checks would be required if fatigue failure is anticipated; in which case the peak stresses need be calculated and comparisons be made to the endurance limit.

Results:

Vessel shell is thick enough to carry the pipe's operating loads even if the entire pressure thrust load is carried by the nozzle. (Test this by running the analysis with and without pressure thrust.)

## Task 5: Document the analysis.

Input echo
Plot
Output report
Annotated stress isometric
File backup